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POSSIBILITIES OF REMOVING CONDENSATE FROM A HEAT RECOVERY UNIT UTILIZABLE IN PAPER INDUSTRY

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ABSTRACT

Methods, processes and equipments currently used for heat recovery systems are very diverse in different branches of industry including paper industry. A very important process applied in heat recovery units is condensate removal from the heat recovery units because of optimization of the heat recovery process and extending the working life of heat recovery units. Using of heat recovery units with condensate removal in paper industry fits the innovation trends and means heat energy saving that can be realized by increase of heat recovery efficiency. Heat recovery system with condensate removal should be installed near a drying cover of a paper machine due to reduction of heat loss and pipeline length. Integration of designed spiral heat recovery unit with condensate removal into the existing dryer section of paper machine in a paper mill will lead to decrease of heat consumption and increase heat recovery efficiency up to 91.7 %.

KEYWORDS: Heat recovery, heat recovery efficiency, heat recovery unit, paper industry, paper machine, water condensate removal.

INTRODUCTION

Condensed steam or liquid condensate forms in many heat recovery systems which use steam or mixture of steam and air as heat media. Formed by condensed steam, liquid condensate should be drained from pipelines and equipments, also from installed heat recovery unit on a continuous basis to avoid the risk of water hammer and to maintain heat exchange efficiency in processes. Normally, condensate contains around 25 % of the usable energy of the steam from which it formed. Using condensate to heat the boiler feedwater or tray water in production of paper results in lower water to steam conversion or it contributes to more effective production of paper. In other words, less fuel is required to produce steam from hot water rather than cold water. As a rule of thumb: every 6°C rise in feedwater temperature achieves by using "free" energy from hot liquid condensate equates approximately to a 1 % fuel saving. It is also known that improvement of drainage depends on increasing temperature of pulp slurry. In fact, liquid condensate is distilled water with a little amount of total dissolved solids. More condensate returned to the feedtank or production process reduces the need for boiler blowdown, which is used to reduce the concentration of dissolved solids in the boiler and in the whole production process. There are many analytical and experimental investigations in determining the condensate amount and in method of removing condensate from systems, however, it concerns mostly air conditioning (Al-Farayedhi et al. 2014). In drying process in the frame of production of paper, electricity or steam is used to heat the cylinders with various specific dimensions and deployment (Åkesson and Slätteke 2006). The cylinders are located in paper machine drying section and the heating process mediated through the cylinders causes the separation of steam from pulp slurry. Thus, reduction of total costs in paper mills as large industrial consumer of energy is linked to two industrial solutions for the most effective reduction of energy consumption: cogeneration units by production of heat and electricity (Qu et al. 2014) and heat recovery systems with high heat recovery efficiency (Wallin and Claesson 2014). Heat recovery system includes a heat recovery unit and ventilation pipelines joining the heat recovery unit with paper machine drying section. A very good solution of the issue how to remove the condensate from the heat recovery unit is installation of condensate pump with electric motor (Reimann et al. 2003) or functional specific U-tube which removes the condensate automatically – without direct use of pump with electric motor in the equipment (Pažitný et al. 2016). High interest is in the research field of spiral heat exchangers with additional equipments because such types of exchangers ensure the entire countercurrent flow and the heat exchange surfaces will redouble in the heat exchange measuring unit (Roetzel and Spang 2010) and the suitable design of additional equipment such as functional specific U-tube to remove condensate can be done for such type of exchanger. An extensive research has been realized for evaluating the influence of channel hydraulic diameter, internal dimensions, atmospheric pressure, velocity of airflow in spiral heat exchanger and amount of withdrawn condensate on heat recovery efficiency, in order to design the parameters of the spiral heat exchangers for utilization in chosen paper mill. The main objective of this study is to evaluate the possibilities of effective condensate removal which is joined to the spiral heat exchanger – heat recovery unit, according to the horizontal temperature profile of operating paper machine drying section, in order to design the dimensions of heat transfer area in view of the connection with functional specific U-tube for condensate removal. The main reason for that design is large heat recovery efficiency after application of the heat recovery unit with functional specific U-tube for condensate removal in conditions of our pilot paper machine – approximately 99 %.

MATERIAL AND METHODS

Materials

Every auxiliary material used in the construction of heat recovery unit and connected functional specific U-tube was commercially available. An aluminium spiral shaped plate with 0.5 mm thickness and dimensions of heat transfer formed area of 460 mm x 22,720 mm was used as a heat exchange wall in the heat recovery unit. The stated plate was a crucial part of the equipment (Pažitný et al. 2013). The external planking was water-resistant with rubber lining and air tightness was ensured by polyethylene sheet. Hot air containing hot steam as a low-potential heat source was provided by operating paper machine drying section (year of recommissioning:

1998; country of origin: Germany). The operating paper machine included multi-cylinder drying section with 20 cylinders. Cold air was pumped from the paper mill facility. Additionally, the heat recovery system was also formed by two pipelines (2 x 10 m) which consisted of two aluminium layers with internal insulation made of mineral wool embedded between them. The internal diameter of the heat exchange system pipes was 0.1 m and also every inputs and outputs of the heat recovery unit had circular section with mentioned internal diameter. In view of the base of the heat recovery unit, the free terminal part of the designed functional specific U-tube was lying in the height ranging from 0.205 m to 0.325 m above the base (Pažitný et al. 2016).

Methods

Location and delimitation of the heat recovery unit was determined according to the horizontal temperature profile of the operating paper machine drying section. Measurement of the temperature values was realized by infrared thermometer (HD 500, Extech Instruments Corporation, USA). Velocity values of actual currents of hot and cold air for design or calculations of dimensions of the functional specific U-tube connected to the spiral heat recovery unit were measured by laboratory anemometer (GM 816, Sinokit Enterprise Limited, China). Other data for heat recovery efficiency determination of the heat recovery unit, such as the actual temperature and relative humidity of the currents of hot and cold air – input and output, were collected by electronic equipment (KlimaLogg Pro Cat. No. 30.3039.IT, Elso Philips Service JSC, SR). Data were gathered by four specific sensors located in the heat recovery unit and transferred by local wireless network specified under IEEE 802.11: 2012 to computational device with signal receiver of the wireless network. The processed data from the computational device were further transferred to a computer using an included USB wireless transceiver (Cat. No. 30.3175, Elso Philips Service JSC, SR). The collected data were recorded every 5 minutes and they were processed by software KlimaLogg Pro (TFA Dostmann LLP, Wertheim-Reicholzheim, Germany). The actual values of atmospheric pressure in the environment of paper mill for December 2015 were obtained by portable device (Weather Station, type TE688NL, EMOS LLC, Czech Republic). The amount of withdrawn condensate was measured by suitable graduated cylinders.

RESULTS AND DISCUSSION

Horizontal temperature profile of operating paper machine drying section

Specification of the temperature profile of operating paper machine drying section was realized by temperature measurements of individual cylinders which were forming the main part of the multi-cylinder dryer section of the operating paper machine. The resulted values of temperature for specification of the temperature profile of operating paper machine drying section were measured in June 2013 and December 2015 (Tab. 1). The paper mill was just working with cogeneration at the end of June 2013 and the results concerning the temperature profile also include the measurements of temperature values without utilization of cogeneration realized at the beginning of June 2013. The obtained results show that the most suitable location of terminal part of pipeline connected to the heat recovery unit input with the most steam concentration was in the middle region of the multi-cylinder dryer section of the operating paper machine. The measurements of heat recovery efficiency were realized at that location of the terminal part of pipeline connected to the heat recovery unit. However, the higher temperature values were found out on the cylinder 18 (133.4°C, 130.6°C or 117.4°C, Tab. 1) but drainage of produced paper was greater in that location, thus the location of the terminal part of pipeline over mentioned cylinder was not suitable. Additionally, temperature values depended on the year period.

Tab.1: Specification of horizontal temperature profile of operating paper machine drying section obtained in 2013 and 2015.

Cylinder rank of operating paper machine	Average temperature measured in June 2013 / without cogeneration (°C)	Average temperature measured in June 2013 / with cogeneration (°C)	Average temperature measured in December 2015 / with cogeneration (°C)
Cylinder 1	40.3	53.2	45.0
Cylinder 2	39.6	45.3	45.8
Cylinder 3	38.5	49.7	58.0
Cylinder 4	39.6	46.8	80.8
Cylinder 5	38.9	48.6	57.0
Cylinder 6	46.9	66.7	83.7
Cylinder 7	59.7	59.4	64.2
Cylinder 8	56.1	52.7	82.0
Cylinder 9	50.1	77.1	96.5
Cylinder 10	49.4	57.9	66.3
Cylinder 11	103.2	116.4	86.9
Cylinder 12	102.5	92.3	100.8
Cylinder 13	104.0	110.6	106.7
Cylinder 14	108.0	119.2	103.9
Cylinder 15	107.0	120.4	91.0
Cylinder 16	113.5	121.9	102.0
Cylinder 17	114.3	95.2	71.4
Cylinder 18	133.4	130.6	117.4
Cylinder 19	94.3	90.5	70.2
Cylinder 20	52.0	66.4	57.5

Application of cogeneration in paper mill was very effective also as average temperature of tray water 30.1°C was higher compared to 14.2°C without application of cogeneration although in December average temperature was 16.1°C in the time when cogeneration was also used. This also suggests that temperature values depended strongly on the year period. Cogeneration system in paper mill uses cogeneration unit with high performance for production of heat and electric energy. The cogeneration unit produces heat and electricity by natural gas combustion. As the results show (Tab. 1), using of cogeneration promoted enhancement of average temperature values almost in the entire horizontal profile of multi-cylinder dryer section of the operating paper machine. However, in both cases – in production of paper with or without cogeneration system, the most suitable location of terminal part of pipeline connected to the heat recovery unit was in the middle region of operating paper machine drying section. The technical possibilities of the drying cover of the operating paper machine allowed the location of terminal part of pipeline connected to the heat recovery unit above the cylinder 13. The average temperature in the region between the drying cover of the paper machine and the thirteenth drying cylinder of the operating paper machine drying section was ranging from 48.7°C to 54.7°C. In that case, the cogeneration was also helpful because otherwise the average temperature value would have been 41.0°C. It is important to note that the maximum output of the installed cogeneration unit was 400 kWh, the maximum of heat output was about 340 kWh (average of 85 % of maximum output) and the residual energy was used for electricity production. Incurred losses of energy were included too in the residual energy. The high ratio of total energy transformed to heat energy is very important because the paper production process is essentially drying process with a very high energy demand where a diluted water suspension of pulp slurry containing solid fibre with concentration ranging from less than 0.5 % to 5.0 % (Hund and Barriere 2016) is used. Additionally, water is a material with large value of the specific heat of vaporization and it is known that liquids with high specific heat of vaporization are vaporized with high supplied heat energy compared to liquids with lower specific heat of vaporization. The low-potential heat energy is the highest potential energy waste which could be vented out into the atmosphere or to heat recovery unit (Ghosh 2011). Our designed solution for such total energy saving and heat energy conservation in paper mill is spiral heat recovery unit with countercurrent heat exchange with designed functional specific U-tube which removes the condensate automatically.

Analysis of measuring of airflow velocity

Airflow velocity values measured by laboratory anemometer (GM 816, Sinokit Enterprise Limited, China) at standard laboratory conditions (a total pressure of 101 300 Pa, a total temperature of 15°C, a relative humidity of 60 %) of heat recovery unit environment (ISO 3977-2) because of the correct adjustment of the airflow velocity in the heat recovery unit and correct calculation of heat recovery efficiency values. As shown in Fig. 1, the maximum airflow velocity value (4.5 $m.s^{-1}$) belongs to the heat recovery efficiency value of 92.4 %. However, the maximum heat recovery efficiency value (97.0 %) was achieved at the airflow velocity value of 3.6 $m.s^{-1}$. Thus the heat recovery system consisting of heat recovery unit and insulated pipelines had to work with the airflow velocity value approximately 3.6 $m.s^{-1}$ in the operating conditions. Our efforts were also aimed to maintain the stable airfow velocity in the heat recovery system just at airflow velocity value of 3.6 $m.s^{-1}$ in order to obtain useful and relevant results of measured parameters and optimum heat recovery efficiency values.

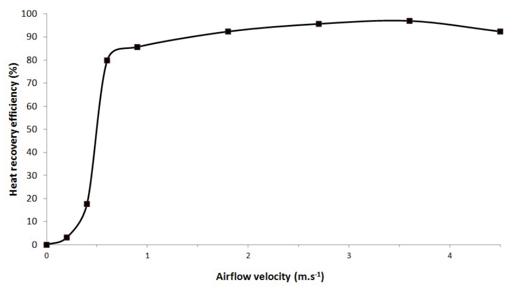

Fig. 1: Dependence of airflow velocity on heat recovery efficiency of spiral heat recovery unit with heat recovery efficiency value maximum at airflow velocity value 3.6 $m{\cdot}s^{-1}$.

Calculation of parameters of humid air flowing from operating paper machine drying cover

The parameters of humid air flowing from drying cover of operating paper machine such as density of humid air, specific heat capacity of the humid air were calculated on base of the automatically measured and collected values relative humidity and temperature of an air-water mixture and on base of the atmospheric pressure which had long-term average value 100 567 Pa for December 2015. The Eq. 1 for specific calculation of humid air density was used according to the findings and derivations in our similar previous study related to the field of heat recovery systems (Pažitný et al. 2015):

$$\varsigma = \frac{1.316\ x\ 10^{-3}}{T\ (K)} \left(2.65\ p - 133.3\ \varphi\ x\ e^{\left(18.3036 - \frac{3816.44\ K}{T\ (K) - 46.13\ K}\right)}\right)\ [kg.m^{-3}] \tag{1}$$

Calculation of heat recovery efficiency of heat recovery unit

Calculation of heat recovery efficiency of a heat recovery unit (Fig. 2) dimensioned pursuant to the previous papers (Picón-Núñez et al. 2007; Picón-Núñez et al. 2009; Picón-Núñez et al. 2012) is based on the known parameters of humid air including the density calculated by using the Eq. 1. The heat recovery efficiency calculation is based on the temperature and relative humidity values of the hot and cold air flowing in the introduced countercurrent heat recovery unit. The values were measured, recorded and collected by the system of four sensors located on input and output of this equipment and the computational device with signal receiver of the wireless network. The heat recovery unit was used in heat recovery system at operating paper machine, thus it is necessary to calculate the value of specific heat capacity of the humid air by following Eq. 2 (Pažitný et al. 2015):

$$c_p = \frac{p_{Ao}}{p - p_{Ao}} \frac{MW_A}{MW_{DA}} c_{pA} + c_{pDA} \tag{2}$$

where: c_{pA} - specific heat capacity of water vapour (J.kg^{-1}.K^{-1}),
c_{pDA} - specific heat capacity of dry air (J.kg^{-1}.K^{-1}).

Dependence of specific heat capacity of pure water vapour and dry air on temperature is not significant in the wide scope. The specific heat capacity of pure water vapour (c_{pA}) for temperature ranging from 0 to 80°C is equal to 1.93 J.kg^{-1}.K^{-1}. The specific heat capacity of dry air (c_{pDA}) for the same temperature range is equal to 1.01 J.kg^{-1}.K^{-1}.

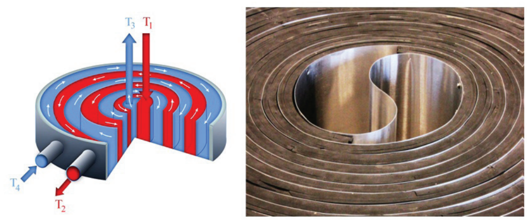

Fig. 2: Spiral heat recovery unit (Pažitný et al. 2015) with depicted countercurrent flow of two currents (left) – hot air current (red colour) and cold current (blue colour); detail of spiral in the heat recovery unit (right).

Heat flow rate of hot and cold air current depended on different mass flow rate value and specific heat capacity value if the heat recovery system is used in operating conditions. Thus, resulting mathematical relationship – Eq. 3 can be applied for the heat recovery efficiency calculation for introduced spiral heat recovery unit in operating conditions:

$$\eta_{SHRU} = \frac{\tilde{v}_{cac}\, S\, \tilde{\varsigma}_{cac}\, \tilde{c}_{pcac}\, (T_3 - T_4)}{\tilde{v}_{hac}\, S\, \tilde{\varsigma}_{hac}\, \tilde{c}_{phac}\, (T_1 - T_2)}\ x\ 100\ \% \tag{3}$$

where: $\tilde{v}_{cac}$ – average velocity of cold humid air current measured on input or output to the spiral heat recovery unit (m.s^{-1}),
$\tilde{\varsigma}_{cac}$ – average density of cold humid air current calculated by using the Eq. 1. (kg.m^{-3}),
$\tilde{c}_{pcac}$ – average specific heat capacity of cold humid air current calculated by using the Eq. 2 (J.kg^{-1}.K^{-1}),

$\tilde{v}_{hac}$ – average velocity of hot humid air current measured on input or output to the spiral heat recovery unit (m.s^{-1}),
$\tilde{\varsigma}_{hac}$ – average density of hot humid air current calculated by using the Eq. 1 (kg.m^{-3}),
$\tilde{c}_{phac}$ – average specific heat capacity of hot humid air current calculated by using the Eq. 2 (J.kg^{-1}.K^{-1}),
T_1 – thermodynamic temperature of hot humid air current recorded on input into the spiral heat recovery unit (K),
T_2 – thermodynamic temperature of hot humid air current recorded on output from the spiral heat recovery unit (K),
T_3 – thermodynamic temperature of cold humid air current recorded on output from the spiral heat recovery unit (K),
T_4 – thermodynamic temperature of cold humid air current recorded on input into the spiral heat recovery unit (K),
S – area of pipe cross-section on air currents input or output of the spiral heat recovery unit in m^2 – used pipes with internal diameter 0.1 m had cross-section 7.85x10^{-3} m^2.

In general, for thermodynamic temperature is known relationship:

$$T(K) = T(°C) + 273,15 \tag{4}$$

where: T(K) - thermodynamic temperature (K),
T(°C) - Celsius temperature measured in degrees Celsius (°C).

It was found that values of average specific heat capacity of both humid hot and cold air current for each measurement were almost constant (1.01 J.kg^{-1}.K^{-1}) and equal (c_{pcac} = c_{phac} = 1.01 J.kg^{-1}.K^{-1}). As area of pipe cross-section on air currents input or output of the spiral heat recovery unit was equal in all cases, the heat recovery efficiency was calculated according to the following Eq. 5:

$$\eta_{SHRU} = \frac{\tilde{v}_{cac}\,\varsigma_{cac}\,(T_3 - T_4)}{\tilde{v}_{hac}\,\varsigma_{hac}\,(T_1 - T_2)}\ x\ 100\ \% \tag{5}$$

Tab. 2 shows the calculated values of heat recovery efficiency and amounts of withdrawn condensate. The calculation of the heat recovery efficiency values is based on the average velocity values and other quantities by using the Eq. 5. It was found that the heat recovery system was stabilized until 60th minute at similar trend as in our previous study referring about laboratory and pilot plant testing of the heat recovery (Pažitný et al. 2013; Pažitný et al. 2015). However, heat recovery efficiency was stabilized after 60th minute and its values varied between 91.24 % and 91.70 %.

We expect that low stability of the heat recovery system was caused by the cyclic production of paper and unstable heat supply in the frame of operating paper machine. However, the heat recovery efficiency of the heat recovery unit reached high level (91.70 %) which is much higher value than that achieved by heat recovery units routinely used in operating paper machines. As shown in Tab. 2, the total condensate amount removed during the chosen period of production of paper in operating conditions was 1132 cm^3. It is interesting finding that condensate formation was stopped just in 60th minute of the chosen period of the introduced paper production. We suppose that it is related to stabilizing of heat recovery unit and also the whole heat recovery system after 60th minute. The results showed that it is possible to remove condensate formed from steam during paper production in operating conditions more effectively without negative effect to heat recovery efficiency value. Vice versa, heat recovery efficiency due to the condensate removal increased rapidly.

Tab. 2: Calculated average thermodynamic quantities of hot air current and cold air current and heat recovery efficiency of heat recovery unit.

Time (min)	Heat recovery efficiency (%)	Quantity of withdrawn condensate (cm^3)
0	0.00	0.00
5	1.05	12.00
10	2.11	42.00
15	3.55	68.00
20	7.15	135.00
25	12.55	220.00
30	17.82	451.00
35	24.18	801.00
40	34.02	955.00
45	47.77	1037.00
50	71.12	1100.00
55	88.85	1120.00
60	90.10	1132.00
65	91.02	1132.00
70	91.24	1132.00
75	91.51	1132.00
80	91.70	1132.00
85	91.55	1132.00
90	91.62	1132.00
95	91.52	1132.00
100	91.49	1132.00
105	91.45	1132.00
110	91.44	1132.00
115	91.44	1132.00
120	91.42	1132.00
125	91.38	1132.00
130	91.40	1132.00
135	91.41	1132.00
140	91.43	1132.00
145	91.42	1132.00
150	91.43	1132.00

CONCLUSIONS

Heat recovery unit with connected functional specific U-tube was used for the study of possibilities of condensate removing from a heat recovery unit, thus also from the heat recovery system consisting of the introduced heat recovery unit and insulated pipelines. Heat recovery efficiency of heat recovery unit was tested in conditions of operating paper machine. The humid hot air was withdrawn from the drying cover of the operating paper machine and the humid cold air was withdrawn from the paper mill environment. Both currents were led through the pipeline system and they met in the introduced heat recovery unit with countercurrent flow of the currents. This was followed by expected heat exchange in heat recovery unit with heat recovery efficiency increase.

The calculated heat recovery efficiency depended on change of parameters of both humid air currents. The monitored values of parameters such as the airflow velocity, the relative humidity and thermodynamic temperature of both humid hot and humid cold air currents were used to calculate the density and the average specific heat capacity of both humid air currents. Additionally, density and specific heat capacity was also calculated based on atmospheric pressure which had long-term average value 100 567 Pa obtained in December 2015. For each measurement the average specific heat capacity values were almost constant (1.01 $J.kg^{-1}.K^{-1}$) and they were equal for both humid hot and humid cold air current. Changes in density values for both humid hot and humid cold air currents were negligible.

The introduced heat recovery unit with heat recovery efficiency with maximum value of 91.70 % and also heat recovery units with proportional dimensions can be applied in paper industry because of high heat consumption of paper mills. Additionally, the solution of condensate removing through functional specific U-tube which removes the condensate from heat recovery system automatically contributes to designs of maintenance-free heat recovery equipments in various industrial branches with high heat energy demand. However, the plants should meet one essential condition – they should produce hot steam or a mixture of hot air and hot steam analogical to operating paper machine drying section.

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REFERENCES

1. Al-Farayedhi, A.A., Ibrahim, N.I., Gandhidasan, P., 2014: Condensate as a water source from vapour compression systems in hot and humid regions, Desalination 349: 60-67.
2. Åkesson, J., Slätteke, O., 2006: Modeling, calibration and control of a paper machine dryer section (ed. The Modelica Association), In: Modelica 2006, September 4th – 5th. Pp 411-420.
3. Ghosh, A.K., 2011: Fundamentals of paper drying – theory and application from industrial perspective, In: Evaporation, condensation and heat transfer, (ed. Amimul Ahsan). Pp 535-583. In Tech Europe, Rijeka, Croatia.
4. Hund, R., Barriere, C., 2016: Nouveau procede de fabrication de papier mettant en oeuvre un copolymere base ayant reagi avec un aldehyde comme agent de resistance a sec, de retention, d'egouttage et de machinabilite. (Novel method for producing paper implementing a base copolymer having reacted with an aldehyde as a dry strength, retention, drainage and machinability agent), Published by WIPO in one of the EPO official languages on September 6th, 2013; EP2820188 B1 (in French).

5. ISO 3977-2, 1997: Gas turbines - Procurement - Part 2: Standard reference conditions and ratings.
6. Pažitný, A., Boháček, Š., Medo, P., 2013: Vývoj nových technológií umožňujúcich rekuperáciu tepla a ich aplikácia v papierenskom priemysle, (Development of new technologies for heat recovery and their application in paper industry), Energetika 63(8-9): 481-483 (in Slovak).
7. Pažitný, A., Boháček, Š., Medo, P., Balberčák, J., 2015: Application of innovative heat recovery unit in paper industry with potential utilization in wood-processing and furniture industry, Wood Research 60(1): 101-112.
8. Pažitný, A., Boháček, Š., Medo, P., Brezániová, Z., Russ, A., Stankovská, M., Ihnát, V., Schwartz, J., Gigac, J., Balberčák, J., 2016: Rekuperačné zariadenie na báze špirály s integráciou odvádzania kondenzátu U-trubicou, (Heat recovery unit based on a spiral with the integration of U-tube condensate removal). Issued by the Industrial Property, Office of the Slovak Republic on June 28th, 2016; PP 50043-2016 (in Slovak).
9. Picón-Núñez, M., Canizalez-Dávalos, L., Martínez-Rodríguez, G., Polley, G.T., 2007: Shortcut design approach for spiral heat exchangers, Food and Bioproducts Processing 85(C4): 322-327.
10. Picón-Núñez, M., Canizalez-Dávalos, L., Medina-Flores, J.M., 2009: Alternative sizing methodology for compact heat exchangers of the spiral type, Heat Transfer Engineering 30(9): 744-750.
11. Picón-Núñez, M., Polley G.T., Riesco-Ávila, J.M., 2012: Design space for the sizing and selection of heat exchangers of the compact type, Chemical Engineering Transactions 29: 217-222.
12. Qu, M., Abdelaziz, O., Yin, H., 2014: New configurations of a heat recovery absorption heat pump integrated with a natural gas boiler for boiler efficiency improvement, Energy Conversion and Management 87: 175-184.
13. Roetzel, W., Spang, W., 2010: C1 Thermal design of heat exchangers, In: VDI heat atlas, second edition (ed. VDI Gesellschaft Verfahrenstechnik und Chemieingenieurwesen, Düsseldorf, Germany). Pp 33-66. Springer-Verlag, Berlin Heidelberg.
14. Reimann, R., Hille, A., Czechowicz, B., 2003: Condensate pump for rooftop air conditioning unit. Issued by the US Patent & Trademark Office on May 5th, 2003; US20040221606 A1.
15. Wallin, J., Claesson, J., 2014: Analyzing the efficiency of a heat pump assisted drain water heat recovery system that uses a vertical inline heat exchanger, Sustainable Energy Technologies and Assessments 8: 109-119.

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